CONTROL VALVE SYSTEM

CROSS-REFERENCE TO THE RELATED ART

This nonprovisional application claims priority under 35 U.S.C. § 119(a) on Patent Application No. 2003-040445 filed in Japan on February 19, 2003, the entire contents of which are hereby incorporated by reference.

BACKGROUND OF THE INVENTION

Field of the Invention

The present invention relates to a control valve system of a variable displacement swash plate type compressor for use in a heating and cooling air conditioner.

Description of the Related Art

Variable displacement swash plate type compressors are designed to adjust the pressure in the crank chamber to thereby control the discharge capacity. For example, in a variable displacement swash plate type compressor for use in a heating and cooling air conditioner, the pressure in the crank chamber is autonomously adjusted, so that a differential pressure between predetermined two points in a refrigerating circuit approaches a target differential pressure that is determined based on external information supplied from an external information detecting means. In other words, the differential pressure between these two points and, by extension, the discharge capacity, is feedback controlled.

Japanese provisional patent publication no. 2001-107854 discloses a control valve system of a variable capacity swash plate type compressor for autonomously adjusting the pressure in the crank chamber. This control valve system is provided with a valve that is variable in

opening and that has a valve body arranged to be urged in one direction by an electromagnetic force corresponding to a target differential pressure between the predetermined two points in the refrigerating circuit and to be urged in the reverse direction by an actual differential pressure therebetween, the target differential pressure being determined based on external information supplied from external information detecting means. The control valve system is further designed to introduce the discharge gas to the crank chamber through the aforementioned valve for autonomous adjustment of the pressure in the crank chamber, in which adjustment the differential pressure between the two points and, by extension, the discharge capacity, is feedback controlled, so that the differential pressure approaches the target differential pressure.

The control valve system disclosed in JP-2001-107854 A requires that the differential pressure between the predetermined two points in the refrigerating circuit be increased, in order to achieve a stable feedback control of the differential pressure. To this end, for example, a restrictor must be provided between these two points.

However, the control valve system using a restrictor poses a problem that, if the degree of restriction is made large, a pressure loss due to the restriction increases with the increasing discharge capacity, resulting in a low compressor efficiency. On the other hand, if the degree of restriction is made small, the differential pressure between the two points decreases with the decrease in discharge capacity, which makes a stable feedback control of the differential pressure difficult, making it difficult to stably perform the feedback control of the discharge capacity.

SUMMARY OF THE INVENTION

The object of the present invention is to provide a control valve system of a variable capacity swash plate type compressor, which is capable of performing a stable feedback control of the discharge capacity in a range from a small discharge capacity to a large discharge capacity, and capable of suppressing the decrease in compressor efficiency at large discharge capacity.

The present invention provides a control valve system of a variable displacement swash plate type compressor for use in a heating and cooling air conditioner, which comprises a throttling valve provided in a refrigerating circuit; a constant differential pressure valve arranged to open when a differential pressure between upstream and downstream pressures of the throttling valve reaches a predetermined value, thereby introducing compressor discharge gas to a crank chamber; external information detecting means for detecting external information such as cooling load or vehicle running state; and control means for determining an opening of the throttling valve based on the external information.

In the control valve system of this invention, a target quantity of flow of refrigerant passing through the throttling valve and, by extension, a target discharge capacity of the compressor, is determined based on the pressure setting of the constant differential pressure valve and the opening of the throttling valve which is in turn determined based on the external information. The compressor discharge gas is introduced through the constant differential pressure valve, whereby the pressure in the crank chamber is autonomously adjusted. Thus, the differential pressure between the upstream and the downstream pressure of the throttling valve is feedback

controlled so as to approach the pressure setting of the constant differential pressure valve, so that the quantity of flow of the refrigerant passing the throttling valve is feedback controlled to approach the target quantity of flow. Consequently, the discharge capacity of the compressor is feedback controlled to approach the target discharge capacity.

In case that the pressure setting of the constant differential pressure valve is set to an appropriate value, it is possible to stably feedback control the differential pressure between the upstream and the downstream pressure of the throttling valve in a range from a small discharge capacity to a large discharge capacity, thus achieving a stable feedback control of the discharge capacity of the compressor. When the external information indicates the necessity of a large quantity of flow, the opening of the throttling valve is set to a large value to thereby make it possible to eliminate the possibility of reduction in compressor efficiency due to the pressure loss at large discharge capacity.

In this invention, it is preferable that the throttling valve be an electromagnetic valve and integrally mounted to the constant differential pressure valve.

The electromagnetic valve whose opening can be arbitrarily set by means of duty control is suitable to be used as the throttling valve. When the throttling valve is integrally mounted to the constant differential pressure vale, the resultant control valve system can be compact in size.

Preferably, the constant differential pressure valve is arranged to introduce the compressor discharge gas on the upstream side of the throttling valve into the crank chamber.

In an arrangement introducing the compressor discharge gas on the downstream side of the throttling valve into the crank chamber, the discharge gas cannot be introduced into the crank chamber when the air conditioner stops operating and hence the throttling valve is closed. This makes it impossible to reduce the discharge capacity when the air conditioner stops. Such drawback can be eliminated by the just-mentioned preferred embodiment in which the discharge gas on the upstream side of the throttling valve is introduced into the crank chamber.

Preferably, the control valve system is provided with a cutoff valve disposed on the downstream side of the throttling valve.

The provision of the cutoff valve can prevent high pressure gas in the refrigerating circuit from acting on the constant differential pressure valve, when the air conditioner stops operating and the throttling valve is closed. This ensures that the compressor discharge gas on the upstream side of the throttling valve is introduced to the crank chamber, thus positively reducing the discharge capacity when the air conditioner stops.

Preferably, a discharge gas inflow chamber is formed on the upstream side of the throttling valve, and the compressor discharge gas in the discharge gas inflow chamber is introduced into the crank chamber. The discharge gas inflow chamber has an inlet thereof directed tangential to a wall surface of the discharge gas inflow chamber.

In case that the inlet of the discharge gas inflow chamber is directed tangential to a wall surface of the discharge gas inflow chamber, the compressor discharge gas entering the discharge gas inflow chamber makes a circling motion therein, so that lubricating oil contained in the

compressor discharge gas is separated therefrom by means of centrifugal force. The separated lubricating oil is introduced through the constant differential pressure valve into the crank chamber together with the compressor discharge gas, and thus the lubricating oil is positively supplied to the crank chamber.

Preferably, the discharge gas inflow chamber is formed with a plurality of inlets that are circumferentially spaced from one another.

With the discharge gas inflow chamber formed with circumferentially spaced inlets, the compressor discharge gas makes a circling motion in the discharge gas inflow chamber, thus ensuring that the lubricating oil is separated from the compressor discharge gas.

Preferably, the throttling valve has a pressure receiving portion that presses the throttling valve in a direction to be opened when it receives a downstream side pressure.

The throttling valve having such a pressure receiving portion decreases a pressing force due to the downstream side pressure acting on the throttling valve. As a result, the accuracy in controlling the opening of the throttling valve can be improved.

Preferably, the pressure receiving portion has the same area as that of a downstream-side pressure receiving surface of the throttling valve.

In case that the pressure receiving portion and the downstream-side pressure receiving surface of the throttling valve have the same area, a pressing force due to the downstream side pressure acting to open the throttling valve balances a pressing force due to the downstream side pressure acting to close the throttling valve. This makes it possible to carry out an accurate

control of the throttling valve opening.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will become more fully understood from the detailed description given herein below and the accompanying drawings which are given by way of illustration only, and thus, are not limitative of the present invention, and wherein:

Fig. 1 is a block diagram showing a vehicle-mounted air conditioner equipped with a variable displacement swash plate type compressor provided with a control valve system according to an embodiment of this invention;

Fig. 2 is a sectional view showing the control valve system when the air conditioner is in operation; and

Fig. 3 is a sectional view showing the control valve system when the air conditioner stops operating.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

In the following, a control valve system of a variable displacement swash plate type compressor according to one embodiment of this invention will be described.

As shown in Fig. 1, a vehicle-mounted air conditioner A is constituted by a variable displacement swash plate type compressor 1, a condenser 2, an expansion valve 3, and an evaporator 4. The air conditioner A is also provided with a damper 5 for switching air passages between fresh air introduction and inside air circulation, a blower 6, and an air conditioner operation panel 7.

The air conditioner operation panel 7 is equipped with an on-off switch 7a and a temperature setter 7b for the air conditioner A, which are operable by the driver or a front seat passenger. A temperature sensor 4a for detecting an air temperature in the compartment is provided near the

evaporator 4, and various sensors for detecting a vehicle running state, such as vehicle speed sensor, engine rotation speed sensor, throttle opening sensor, etc., are provided in the vehicle, not shown. The on-off switch 7a, temperature setter 7b, temperature sensor 4a, and various sensors for detecting a vehicle running state cooperate with one another to form an external information detecting device 8.

The variable displacement swash plate type compressor 1 comprises a main shaft (not shown) coupled to the automotive engine (not shown) without using a clutch, a swash plate (not shown) mounted to the main shaft so as not to be relatively rotatable but to be variable in inclination angle, a piston (not shown), engaged with the swash plate through a shoe, for a linear reciprocal motion with the rotation of the swash plate, a cylinder bore 1a in which the piston is received for sliding motion, a discharge chamber 1b communicating with the cylinder bore 1a through a discharge valve, a crank chamber 1c accommodating the main shaft and the swash plate, and a suction chamber 1d communicating with the cylinder bore 1a through a suction valve. The crank chamber 1c is communicated with the suction chamber 1d through an orifice hole 1e.

The discharge chamber 1b, the condenser 2, the expansion valve 3, the evaporator 4, and the suction chamber 1d are connected with one another by means of a refrigerating circuit 9.

There is provided a control valve system 10 for controlling the discharge capacity of the compressor 1. The control valve system 10 comprises a throttling valve 11 disposed in the refrigerating circuit 9 near the discharge chamber 1b, a constant differential pressure valve 12

adapted to open to introduce compressor discharge gas into the crank chamber 1c when the differential pressure between an upstream pressure P" and a downstream pressure P' of the throttling valve 11 reaches a predetermined value, the aforementioned external information detecting device 8, a controller 13 for determining the opening of the throttling valve 11 based on external information supplied from the external information detecting device 8, and a driving circuit 14 for the throttling valve 11.

Referring to Fig. 2, the throttling valve 11 and the constant differential pressure valve 12 will be described in detail.

The throttling valve 11 comprises a coil 11a, a stationary iron core 11b, a movable iron core 11c, a rod 11d fixed to the movable core 11c, a valve body 11e fixed to an end portion of the rod 11d, and a valve seat 11f, the coil 11a being connected to the driving circuit 14 through wires, not shown.

On the upstream side of the valve body 11e, an annular discharge gas inflow chamber 11d is provided coaxially with the rod 11. The discharge gas inflow chamber 11g has an outer peripheral wall formed with a plurality of discharge gas inlets 11g' so as to be circumferentially spaced from one another. These discharge gas inlets 11g' are directed perpendicularly to the center axis of the discharge gas inflow chamber 11g and tangential to the inner peripheral surface of the outer peripheral wall of the chamber 11g. The discharge gas inlets 11g' are communicated through the refrigerating circuit 9 to the discharge chamber 1b of the compressor 1.

The discharge gas inflow chamber 11g is communicated with a chamber 11h that is formed on the upstream side of the valve body 11e so as to be adjacent to the valve body

11e. A chamber 11i is formed on the downstream side of the valve body 11e so as to be adjacent thereto, and is communicated with the chamber 11h.

A gas passage 11j extending from the chamber 11i is communicated with a chamber 11k formed behind the chamber 11h. A movable plate 11m, having a first pressure receiving surface 11m' disposed in contact with the chamber 11k, is fixed to the rod 11d. The first pressure receiving surface 11m' has its area that is the same as that of a downstream-side pressure receiving surface 11e' of the valve body 11e. A spring 11n that urges the valve body 11e toward the valve seat 11f is disposed in contact with a second pressure receiving surface 11m" that is disposed on the side opposite the first pressure receiving surface 11m'. The second pressure receiving surface 11m" is adjacent to the chamber 11h via a space in which the spring 11n is received. The area of the second pressure receiving surface 11m" is set to a value that is the same as the area of the upstream-side pressure receiving surface 11e" of the valve 11e. As a result, an urging force, due to the downstream-side pressure P' of the throttling valve 11 applied to the first pressure receiving surface 11m" of the movable plate 11m, acting to move the valve body 11e in the direction away from the valve seat 11f balances an urging force, due to the downstream-side pressure P' of the throttling valve 11 applied to the second pressure receiving surface 11e', acting to move the valve body 11e in the direction toward the valve seat 11f. Also, an urging force, due to the upstream-side pressure P" of the throttling valve 11 applied to the second pressure receiving surface 11m' of the movable plate 11m, acting to move the valve body 11e in the direction toward the valve seat 11f balances an urging force, due to the upstream-side

pressure P" of the throttling valve 11 applied to the upstream-side pressure receiving surface 11e" of the valve body 11e, acting to move the valve body 11e in the direction away from the valve seat 11f.

On the downstream side of the throttling valve 11, a chamber 11p is provided to be adjacent to and communicated with the chamber 11i. The chamber 11p has its outer peripheral wall formed with a discharge gas outlet 11p' that is connected through the refrigerating circuit 9 to the condenser 2 of the air conditioner A.

The cutoff valve 15 is disposed in the chamber 11p and comprises a valve body 15a, a valve seat 15b, and a spring 15c that urges the valve body 15a toward the valve seat 15b.

The constant differential pressure valve 12 comprises a rod 12a, a valve body 12b fixed to the vicinity of one end of the rod 12a, a movable plate 12c fixed to another end of the rod 12a, a valve seat 12d, and a spring 12e disposed in contact with the movable plate 12c and urging the valve body 12 toward the valve seat 12d.

On the upstream side and the downstream side of the valve body 12b, chambers 12f, 12g are formed adjacent to the valve body 12b, respectively. The chamber 12f is in communication with the discharge gas inflow chamber 11g of the throttling valve 11 through a gas passage 12h. The chamber 12f has its outer peripheral wall formed with a discharge gas outlet 12g' that is in communication with the crank chamber 1c of the compressor 1 through a passage 12i.

A chamber 12j is formed that accommodates the movable plate 12c and the spring 12e. A first portion 12j' of the chamber 12f accommodating the spring 12e is communicated through a gas passage 12k with the chamber 11i of the throttling valve 11, whereas a second portion 12j" thereof, facing the first portion 12j' with the movable plate 12c

interposed between these portions, is communicated through a gas passage 12m with the chamber 12f.

To the first portion 12j' of the chamber 12j, the downstream-side pressure P' of the throttling valve 11 is introduced through the chamber 11i and the gas passage 12k, whereas the upstream-side pressure P" of the throttling valve 11 is introduced into the second portion 12j" of the chamber 12j through the discharge gas inflow chamber 11g, the gas passage 12h, the chamber 12f, and the gas passage 12m.

The pressure setting of the constant differential pressure valve 12 is fixed at ΔP. More specifically, the spring constant of the spring 12e is set such that, when the differential pressure between the upstream-side pressure P" of the throttling valve 11 applied from the second portion 12j" of the chamber 12j to the movable plate 12c and the downstream-side pressure P' of the throttling valve 11 applied from the first portion 12j' of the chamber 12j to the movable plate 12c is less than the predetermined value ΔP, the valve body 12b is caused to abut against the valve seat 12d so that the communication between the chambers 12f, 12g is prohibited. On the other hand, when the differential pressure exceeds the predetermined value ΔP, the valve body 12b is allowed to move in the direction away from the valve seat 12d by a distance corresponding to the differential pressure. When the differential pressure is equal to the predetermined value ΔP, the valve body 12b is allowed to move in the direction away from the valve seat 12d by a predetermined distance.

The throttling valve 11, the constant differential pressure valve 12, and the cutoff valve 15 are assembled into one piece.

In the following, the operation of the control valve

system 10 having the above-mentioned construction will be described.

The main shaft, not shown, of the variable displacement swash plate type compressor 1 always rotates by being driven by the automotive engine, not shown.

When the air conditioner A is in operation, the controller 13 determines the discharge capacity Q of the compressor 1 and by extension the target quantity Q of flow of the compressor discharge gas, which is refrigerating gas flowing through the refrigerating circuit 9, on the basis of external information supplied from the external information detecting device 8. The controller 13 also determines the preset opening Θ of the throttling valve 11 from the target quantity Q of flow and the pressure setting ΔP of the constant differential pressure valve 12. Further, the controller 13 operates the driving circuit 14 to carry out a duty control of electric power supplied to the coil 11a of the throttling valve 11. An electromagnetic force is exerted between the magnetized movable and stationary cores 11c, 11b, whereby the movable core 11c is caused to move against the urging force of the spring 11n. Thus, the valve body 11e moves in the direction away from the valve seat 11f, so that the opening of the throttling valve 11 is made equal to the preset opening Θ.

The electromagnetic valve constituted by the coil 11a, stationary core 11b, movable core 11c, rod 11d, valve body 11e and valve seat 11f can have the opening that can be arbitrarily set by means of the duty control. Hence, the electromagnetic valve is suitable for use as the throttling valve 11. An urging force, due to the downstream-side pressure P' applied to the first pressure receiving surface 11m' of the movable plate 11m, acting to move the valve body 11e in the direction away from the valve seat 11f

balances an urging force, due to the downstream-side pressure P' of the throttling valve 11 applied to the downstream-side pressure receiving surface 11e' of the valve body 11e, acting to move the valve body 11e in the direction toward the valve seat 11f. In addition, an urging force, due to the upstream-side pressure P" of the throttling valve 11 applied to the second pressure receiving surface 11m" of the movable plate 11m, acting to move the valve body 11e in the direction toward the valve seat 11f balances an urging force, due to the upstream-side pressure P" of the throttling valve 11 applied to the upstream-side pressure receiving surface 11e" of the valve body 11e, acting to move the valve body 11e in the direction away from the valve seat 11f. Therefore, the opening of the throttling valve 11 is determined depending solely on the relation in size between the electromagnetic force applied from the stationary core 11b to the movable core 11c and the urging force of the spring 11n. This makes it possible to accurately control the opening of the throttling valve 11 by means of the duty control of electric power supplied to the coil 11a.

The compressor discharge gas flows from the discharge chamber 1b, through the refrigerating circuit 9 and the discharge gas inlet 11g', into the discharge gas inflow chamber 11g, and flows into the chamber 11h. Then, the discharge gas passes through a gap between the valve body 11e and the valve seat 11f to enter the chamber 11i, and flows into the chamber 11p. When receiving the dynamic pressure of the compressor discharge gas entering the chamber 11p, the valve body 15a of the cutoff valve 15 is kept apart from the valve seat 15 against the urging force of the spring 15c. In other words, the cutoff valve 15 is kept open. Thus, the compressor discharge gas flowing into

the chamber 11p flows to the condenser 2 through the discharge gas outlet 11p' and the refrigerating circuit 9.

When the differential pressure between the upstream-side pressure P" and the downstream-side pressure P' of the throttling valve 11 is less than the predetermined value ΔP, the valve body 12b is caused to abut against the valve seat 12d to close the constant differential pressure valve 12, so that the communication between the chambers 12f, 12g is prohibited, and the compressor discharge gas in the discharge gas inflow chamber 11g is prevented from flowing into the crank chamber 1c. The gas in the crank chamber 1c is discharged to the suction chamber 1d through the orifice hole 1e, resulting in a reduction in the internal pressure in the crank chamber 1c. As a consequence, the inclination angle of the swash plate, not shown, increases to increase the discharge capacity of the variable displacement swash plate type compressor 1, whereby the quantity of flow of the compressor discharge gas passing through the throttling valve 11 is increased to result in the increased differential pressure between the upstream-side pressure P" and the downstream-side pressure P' of the throttling valve 11.

When the differential pressure between the upstream-side pressure P" and the downstream-side pressure P' of the throttling valve 11 exceeds the predetermined value ΔP, the valve body 12b is apart from the valve seat 12d by a distance corresponding to the differential pressure, to open the constant differential pressure valve 12. Thus, the compressor discharge gas whose quantity of flow corresponds to the distance between the valve body 12b and the valve seat 12d flows from the discharge gas inflow chamber 11g to the crank chamber 1c through the gas passage 12h, chambers 12f, 12g, discharge gas outlet 12g' and

passage 12i. Hence, the quantity of flow of the compressor discharge gas flowing into the crank chamber 1c is greater than the quantity of flow of the gas discharged from the crank chamber 1c to the suction chamber 1d, resulting in the increase in internal pressure in the crank chamber 1c. As the internal pressure in the crank chamber 1c increases, the inclination angle of the swash plate, not shown, decreases. This results in the decrease in discharge capacity of the compressor 1, so that the quantity of flow of the compressor discharge gas passing through the throttling valve 11 decreases, thus decreasing the differential pressure between the upstream-side pressure P" and the downstream-side pressure P' of the throttling valve 11.

When the differential pressure between the upstream-side pressure P" and the downstream-side pressure P' of the throttling valve 11 is equal to the predetermined value ΔP, the valve body 12b is apart from the valve seat 12d by the predetermined distance, thus opening the constant differential pressure valve 12. The quantity of flow of the compressor discharge gas, corresponding to the distance between the valve body 12b and the valve seat 12d, flows from the discharge gas inflow chamber 11g into the crank chamber 1c though the gas passage 12h, chambers 12f, 12g, gas outlet 12g' and passage 12i. Equilibrium is established between the quantity of flow of the compressor discharge gas flowing into the crank chamber 1c and that of the gas discharged from the crank chamber 1c to the suction chamber 1d. Thus, the internal pressure in the crank chamber 1c does not increase and decrease, and the inclination angle of the swash plate, not shown, does not increase and decrease. Consequently, the discharge capacity of the variable displacement swash type compressor

1 does not increase and decrease, and the quantity of flow of the discharge gas passing through the throttling valve 11 does not increase and decrease, so that the differential pressure between the upstream-side pressure P" and the downstream-side pressure P' of the throttling valve does not increase and decrease.

The introduction of the compressor discharge gas to the crank chamber 1c and the prohibition of the introduction are autonomously repeated to autonomously adjust the internal pressure in the crank chamber 1c, whereby the differential pressure between the upstream-side pressure P" and the downstream-side pressure P' of the throttling valve 11 is feedback controlled so as to approach the pressure setting ΔP of the constant differential pressure valve 12. Thus, the quantity of flow of the compressor discharge gas passing through the throttling valve 11 is feedback controlled to approach the target quantity Q of flow, and thus the discharge capacity of the compressor 1 is feedback controlled to approach the target value Q. As a result of the feedback control, if the differential pressure between the upstream-side pressure P" and the downstream-side pressure P' of the throttling valve 11 is equal to the pressure setting ΔP of the constant differential pressure valve 12, the quantity of flow of the compressor discharge gas passing through the throttling valve 11 and determined based on the differential pressure ΔP and the preset opening Θ of the throttling valve 11 becomes equal to the target quantity Q of flow. Thus, the discharge capacity of the compressor 1 becomes equal to the target value Q, and the quantity of flow of the refrigerant flowing through the refrigerating circuit 9 becomes equal to the target quantity of flow, Q. As shown by bold arrow in Fig. 1, appropriate air

conditioning that corresponds to the external information can be achieved when the target flow quantity Q of the refrigerant flows through the condenser 2, the expanding valve 3, and the evaporator 4.

By using the constant differential pressure valve 12 whose pressure setting ΔP is set to the appropriate value, the differential pressure between the upstream-side pressure P" and the downstream-side pressure P' of the throttling valve 11 can be stably feedback controlled in a range from a small discharge capacity to a large discharge capacity, making it possible to achieve a stable feedback control of the flow quantity of the compressor discharge gas passing through the throttling valve 11, and by extension, the discharge capacity of the compressor 1. When the necessity of a large flow quantity is indicated by the external information detected by the external information detecting device 8, the opening of the throttling valve 11 is set to be large, and accordingly, there is no fear of the efficiency of the compressor 1 being lowered due to pressure loss at large discharge capacity.

When the on-off switch 7a is turned off and hence the air conditioner A stops operating, the controller 13 operates the driving circuit 14 so as to stop the power supply to the coil 11a.

Thus, the application of electromagnetic force from the stationary iron core 11b to the movable iron core 11c is prevented, so that the movable core 11c receiving the urging force of the spring 11n is moved in the direction away from the stationary core 11b, whereby the valve body 11e is moved in the direction toward the valve seat 11f and abuts against the valve seat 11f. As a result, as shown in Fig. 3, the throttling valve 11 is closed, so that the

compressor discharge gas is prevented from flowing from the chamber 11i into the chamber 11i and from the chamber 11i into the chamber 11p, whereby the flow of refrigerant in the refrigerating circuit 9 is prevented.

When the air conditioner A stops, the expanding valve 3 is closed, and hence the gas passage between the chamber 11i and the expanding valve 3 is spatially closed. Accordingly, the compressor discharge gas no longer flows into the first portion 12j' of the chamber 12j. Since the compressor 1 is in operation, the compressor discharge gas continues to flow into the second portion 12j" of the chamber 12j. As a result, the differential pressure between the gas pressure applied from the second portion 12j" of the chamber 12 to the movable plate 12c and the gas pressure applied from the first portion 12j' of the chamber 12 greatly exceeds the pressure setting ΔP of the constant differential pressure valve 12, whereby the valve 12 is fully opened. Consequently, the compressor discharge gas flows into the throttling valve 11 and flows through the passage 12i into the crank chamber 1c, and the internal pressure in the crank chamber 1c increases to decrease the inclination angle of the swash plate. As a result, the discharge capacity of the compressor 1 decreases to a minimum value, thus suppressing the waste of energy produced by the automotive engine. Even if the constant differential pressure valve 12 is open when the air conditioner A is rendered inoperative and the throttling valve 11 is closed, the compressor discharge gas flows through the passage 12i into the crank chamber 1 and continues to flow into the second portion 12j" of the chamber 12j. Thus, the constant differential pressure valve 12 is fully opened.

After flowing from the throttling valve 12 into the

crank chamber 1c through the constant differential pressure valve 12, the discharge gas flows through the orifice hole 1e into the suction chamber 1d. Subsequently, as shown by bold double arrow in Fig. 1, the discharge gas is sucked from the suction chamber 1d into the cylinder bore 1a of the compressor 1 which is kept in operation, is then discharged from the cylinder bore 1a to the discharge chamber 1b, and is returned to the throttling valve 11.

When the throttling valve 11 is closed, the discharge gas flowing from the chamber 11i into the chamber 11p stops applying a dynamic pressure onto the valve body 15a. The valve body 15a is moved toward the valve seat 15b by means of the urging force of the spring 15c, and abuts against the valve seat 15b, whereby the cutoff valve 15 is closed.

Since the compressor discharge gas flowing into the throttling valve 11 is recirculated to the valve 11 by way of the suction chamber 1d, the gas pressure in the discharge gas inflow chamber 11g rapidly decreases to the vicinity of the suction pressure. In case that the cutoff valve 15 is not closed with the closure of the throttling valve 11, the pressure setting ΔP of the constant differential pressure valve 12 is not exceeded by the differential pressure between the internal pressure in the second portion 12j" of the chamber 12j that decreases with the decreasing gas pressure in the discharge gas inflow chamber 11g and the internal pressure in the first portion 12j' of the chamber 12j that cooperates with the gas passage extending between the chamber 11i and the expansion valve 3 to form a closed space. Thus, the constant differential pressure valve 12 is closed. As a result, the compressor discharge gas is impeded from flowing into the crank chamber 1c to inhibit the rise in the internal pressure in the crank chamber 1c, whereby the inclination

angle of the swash plate is inhibited from decreasing, and the minimization of the discharge capacity of the variable displacement swash plate type compressor 1 is inhibited. Consequently, there occurs a drawback that the suppression of waste of energy produced by the automotive engine can be impaired. In case that the cutoff valve 15 is closed with the closure of the throttling valve 11, on the other hand, the movable plate 12c is prevented from moving up to a position where it closes the constant differential pressure valve 12, even if the pressure setting ΔP of the valve 12 is not exceeded by the differential pressure between the internal pressures in the first and second portions 12j', 12j" of the chamber 12j, because the volume of a closed space is small that is defined by the chamber 11i, the discharge gas passage 12k, and the first portion 12j' of the chamber 12i. When the movable plate 12c moves in the direction to close the constant differential pressure valve 12, the volume of the first portion 12j' of the chamber 12j increases, and the volume of the closed space defined by the chamber 11i, the discharge gas passage 12k, and the first portion 12j' of the chamber 12j increases. Since the volume of the closed space is small, a slight increase in the volume of the first portion 12j' of the chamber 12j results in a large rate of volume increase of the closed space, so that the internal pressure in the closed space greatly decreases. As a consequence, the differential pressure between the internal pressures in the first and second portions 12j', 12j" of the chamber 12j increases to exceed the pressure setting ΔP of the constant differential pressure valve 12, whereby the movable plate 12c is pushed back in the direction to open the valve 12. Since the valve 12 is kept open, the compressor discharge gas flows into the crank chamber 1c, and the discharge capacity of

the compressor 1 is minimized, whereby the waste of energy produced by the automotive engine is suppressed.

The throttling valve 11, the constant differential pressure valve 12, and the cutoff valve 15 are assembled into one piece, and accordingly, the control valve system 10 is made compact.

The compressor discharge gas on the upstream side of the throttling valve 11 must be introduced into the crank chamber 1c for the reason that, if the discharge gas on the downstream side of the throttling valve 11 is introduced to the crank chamber 11c, the discharge gas cannot be introduced to the crank chamber 11c, making it impossible to reduce the discharge capacity of the compressor 1 when the air conditioner A stops operating and the throttling valve 11 is closed.

Since the discharge gas inlets 11g' of the discharge gas inflow chamber 11g are directed tangential to the inner wall surface of the chamber 11g, the compressor discharge gas entering the chamber 11g makes a circling motion therein, so that lubricating oil contained in the discharge gas is separated therefrom by means of centrifugal force. The separated lubricating oil is introduced through the constant differential pressure valve 12 into the crank chamber 1c together with the discharge gas. Thus, the lubricating oil is positively supplied to the crank chamber 1c.

The discharge gas inlets 11g' of the discharge gas inflow chamber 11g are circumferentially spaced from one another, and therefore, the compressor discharge gas makes a circling motion in the chamber 11g, which ensures that the lubricating oil is separated from the discharge gas.

In the above, the control valve system 10 according to one embodiment of this invention has been described. This

invention is not limited to the foregoing embodiment, and may be modified variously.

For example, the first pressure receiving surface 11m' of the movable plate 11m is not essentially required to have the same area as that of the downstream-side pressure receiving surface 11e' of the valve body 11e. Also, the second pressure receiving surface 11m" of the movable plate 11m may not have the same area as that of the upstream-side pressure receiving surface 11e" of the valve body 11e. As long as the movable plate 11m is formed with the first and second pressure receiving surfaces 11m', 11m" to which the downstream side pressure and the upstream side pressure of the throttling valve 11 are applied, respectively, pressing forces acting on the throttling valve 11 due to the upstream side pressure and the downstream side pressure of the throttle valve 11 are decreased, thus improving the accuracy of control of the opening of the throttling valve 11.